

TRUCK OCCUPANT PROTECTION CONSIDERATIONS IN HEAVY-VEHICLE CRASHES

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Abstract

This paper estimates the crash forces that can occur in heavy truck crashes. A front-rear collision in which a heavy vehicle runs into the back of another heavy vehicle is considered. This case can be analyzed assuming that the crash is ‘plastic’ – that is both vehicles move together afterwards at the same speed. A real-world crash is an intermediate case between a ‘plastic’ collision and an ‘elastic’ collision. A plastic condition always occurs during a real world crash. The average impact force can be calculated for a plastic collision using energy-momentum methods and the measured crush length. The case of two 64.5t B-double trucks is analysed. Crash decelerations of up to about 11g can occur at relatively low collision speeds as a result of the high masses combined with stiff front and rear under-run protection barriers. The maximum deceleration levels that are assumed in the design rules such as the seat and seatbelt anchor regulations ECE R16 & R17 and the mechanical couplings regulation ECE R55.

Keywords: Truck crashes, impact forces, energy-momentum methods, plastic collisions, elastic collisions, ECE regulations, under-run barriers.

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1. Introduction

This paper estimates the crash forces that could occur in truck-to-truck crashes and considers the safety implications for fixtures to heavy trucks. There are clear trends for Australian (combination) trucks to become heavier and for the front of trucks and rear of trailers to become stiffer as a result of under-run protections being fitted. Consequently crash forces in nose-tail truck crashes are tending to increase.

At least three safety concerns arise for truck-cabin occupants from nose-tail crash forces. High impact forces occur at the cabin mounts and at the mechanical couplings between vehicle parts. Inside the cabin high forces occur at suspension seat anchors. Should any of the fixtures break, parts of the vehicles can become unrestrained with potentially fatal consequences.

Design rules such as the ECE regulations or Australian Design Rules do not adequately consider the road safety implications of heavy truck nose-tail crashes. There are no design-rule strength standards for cabin attachment. Furthermore, mechanical coupling strength requirements are based only on the usual pulling and braking deceleration force levels that occur. Vehicle and coupling manufacturers will have their own design safety factors. However, these may not be adequate for even low-speed heavy truck collisions.

Seat strength and seatbelt anchor strength rules that relate to ECE Regulations are based on an assumed peak deceleration level of 6.6g. The analysis presented here shows that decelerations of 11g or more can occur in relatively low-speed collisions between heavy Australian multi-combination trucks. This work was initiated in consequence of a fatal incident in which the mechanical coupling on a B-double prime-mover broke, which allowed the lead trailer to impact the cabin.

3. Types of Collisions

Figure 1 shows the distinction between 'elastic' and 'plastic' collisions. In the elastic collision no energy is absorbed. In the plastic collision the two vehicles move together after the collision at the same speed. Energy is absorbed during the plastic collision because the structures distort. Real-world collisions produce a response between these two limiting cases.

If the brake and drag forces that may exist during the collision interaction are ignored then momentum is conserved for all types of collisions. This is a reasonable simplification because the crash forces are likely to be at least an order of magnitude greater than the drag forces.

To simply, assume that vehicle 1 is stationary and that vehicle 2 hits vehicle 1 along the longitudinal axis at the initial speed V_{2i} .

For a plastic collision the final speed of vehicles 1 & 2 is:

$$V_{2f} = V_{2i} \cdot M_2 / (M_1 + M_2) = \varepsilon_{p2} KE_{initial} \quad (1)$$

M_1 is the mass of vehicle 1 and M_2 the mass of vehicle 2.

The proportion of the initial kinetic energy that is absorbed during any collision is:

$$\varepsilon = 1 - KE_{final} / KE_{initial} \quad (2)$$

For a plastic collision with vehicle 2 hitting vehicle 1:

$$\varepsilon_{p1} = M_1 / (M_1 + M_2), \quad (3)$$

and for a plastic collision with vehicle 1 hitting vehicle 2:

$$\varepsilon_{p2} = M_2 / (M_1 + M_2). \quad (4)$$

The energy proportions depend only upon the masses of the vehicles and not on the initial speed.

For an elastic collision all the collision energy is returned so the final kinetic energy equals the initial kinetic energy and $\varepsilon_e = 0$. The energy stored in elastic elements during the collision is finally returned.

For a real-world collision permanent distortion occurs. The kinetic energy that is converted into heat as the result of bending and breaking of materials is the *absorbed energy*. During the collision interaction, energy will also be stored in elastic elements as well as deformed elements and hence the instantaneous loss of kinetic energy may exceed the *absorbed energy*. The peak lost kinetic energy is called the *collision energy*. This will be further considered in Section 5.

Figure 2 defines the Crush Length (CL) which is the total distortion distance measured at the height which is judged to have the greatest stiffness. The crush length can be measured after the collision. The stiffness of the front of the prime-mover and the rear of the struck trailer determines the crush length (CL).

4. Crash Force Estimates

The average crash force can be estimated using virtual work as follows:

$$\text{Average impact force} = \text{collision energy} / \text{maximum deflection} \quad (5)$$

The collision energy is not readily calculable for a real collision. It could be estimated using advanced computational techniques that calculate the collision energy. However, this level of analysis is seldom available.

The absorbed energy of a plastic collision can be easily obtained by energy-momentum methods. The maximum deflection is unknown in general except that it slightly exceeds the crush length CL. It is expected that the maximum deflection consists of the permanent distortion distance CL and a further compression arising from elastic energy storage. For a plastic collision the average force estimate is:

$$\text{Average impact force}_{\text{plastic collision}} = \text{absorbed energy} / \text{CL} = \varepsilon_{pl} E_o / \text{CL} \quad (6)$$

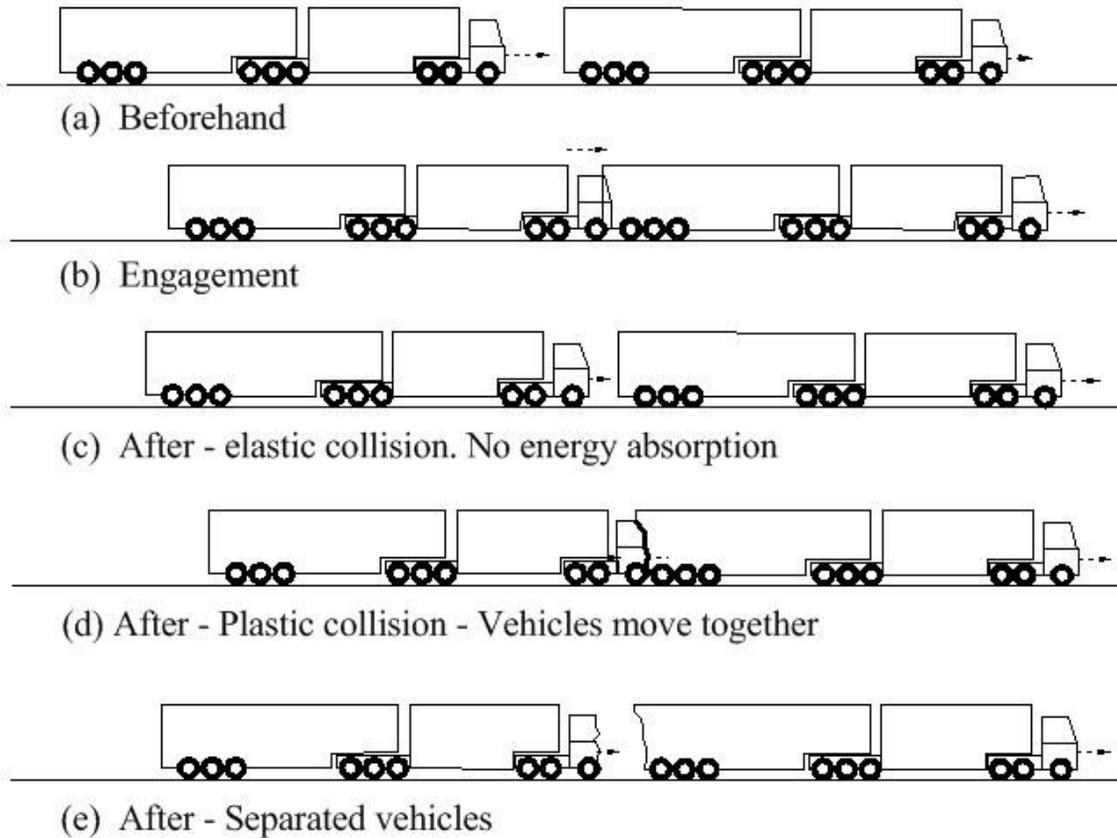


Figure 1 - Collision concepts. Real world crashes produce damaged and separated vehicles (e), which is an intermediate condition between (c) elastic and (d) plastic cases.

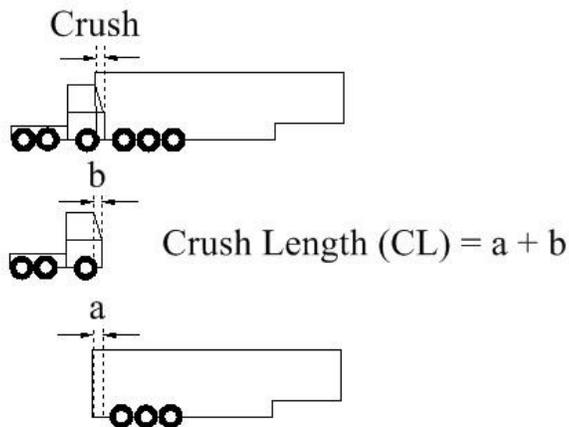


Figure 2 - Crush Length (CL) is the sum of the rear cabin distortion distance and the front trailer distortion distance measured at the stiffest point of the crash

face.

Figure 3 shows the computed averaged impact forces assuming a plastic collision between two heavy trucks (each laden to 64.5t). Equations (2) and (6) have been used. The forces are computed for five assumed crush lengths (0.25, 0.5, 0.75, 1, 1.5m).

The impact speed range in Figure 3 is 0 – 40 km/h which is, relatively low compared to operating speeds. Figure 4 shows that deceleration at the 11g or higher level may occur.

The average forces at rearwards points, such as the couplings, can be estimated based on the known masses that must be decelerated at these points.

The forces that have been calculated for the coupling location on the second truck assume that a rigid connection exists between the collision zone and the coupling location. In fact the truck chassis is flexible to some extent and this will disperse the impact force waveform and thereby reduce the peak force. This is further discussed in Section 7.

The question arising from these considerations then is whether the collision forces of a real-world crash are accurately estimated by the plastic collision estimates. This will be further considered in Section 8.

5. Truck Strength Considerations

It is common engineering practice to design for a safety factor of at least 3. For a mechanical coupling a safety factor of three corresponds to a coupling ultimate strength of 3 x D-value. In the B-double truck example this corresponds to the forces associated with a deceleration of about 2g. The crash impact forces shown in Graph 1 could exceed the usual safety factor levels and the coupling might fail.

The cabin may be rigidly mounted to the chassis or may have a suspension. Considering a cabin with a rear suspension so that the strength is mainly at the front pivots, if the front of the cabin is mounted using two rigid pivots of say 25mm diameter G8.8 steel, then the ultimate shear strength of the front cabin mounts is about $2 \times 491\text{mm}^2 \times 800\text{MPa} \times 0.8 = 314\text{kN}$. If the cabin weighs 1.5t then the force to be transmitted by the cabin mounts due an 11g deceleration is about 162kN. In this example the cabin mounts should survive. For significantly higher deceleration levels this is not true.

In the case of a suspension seat with integral seatbelt anchors the test strength level corresponds to a deceleration level of about 6.6g. The seat is somewhat isolated from the impact forces experienced by the chassis rails because of compliance of the cabin suspension and the cabin structure. However, it is possible that the seat might experience decelerations at the 6.6g level. It is interesting to note that the USA rule FMVSS 208 and the previous Australian seatbelt anchor rule ADR 4/03 specify test forces that are based on an assumed 10g maximum deceleration level.

Impact Forces v Speed with Crush Length as parameter

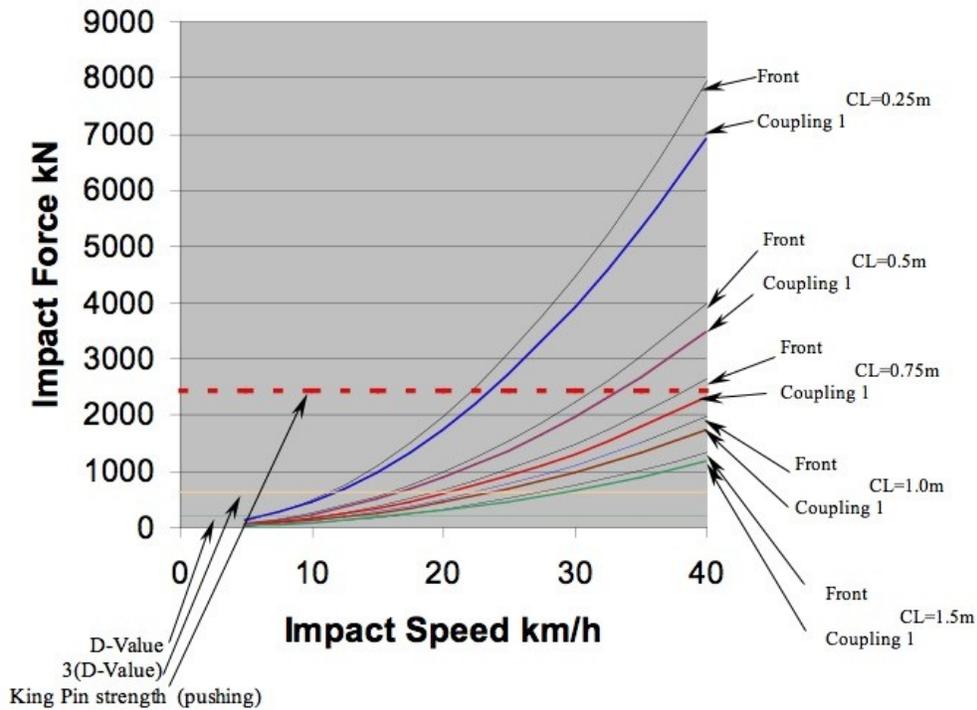


Figure 3 - Forces that occur at the front and at the first coupling on the impacting B-Double in a nose-tail impact assuming a plastic collision. Each truck weighs 64.5t.

Deceleration v Speed for Five Crush Lengths

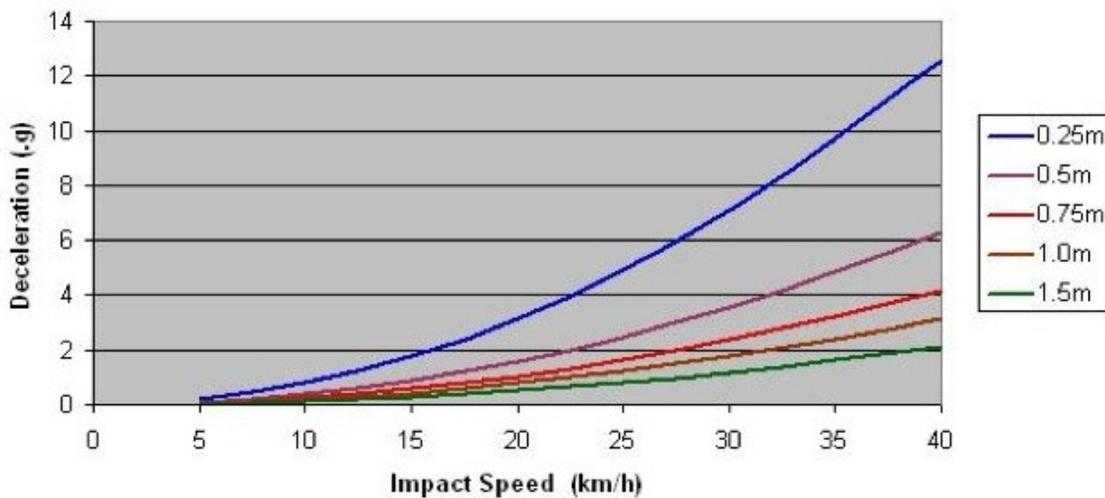


Figure 4 - Computed decelerations for five assumed crush lengths for a plastic collision.

It may be noted that front under-run protection (FUPS) is now commonly fitted to B-double prime-movers in Australia. The requirement is that the FUPS complies with the UN ECE Regulation 93. This requires that the FUPS be able to withstand a force without breaking equal to the weight force of the prime-mover applied horizontally and centrally at the front. In the example above, this force is $\sim 110\text{kN}$ (assuming a prime-mover tare weight of 11t).

6. A Spring Interaction Model

Some insight into the nature of crash forces can be gained by studying ‘elastic’ and ‘plastic’ collisions of two heavy trucks that interact via a linear spring. Figure 5 identifies the situation. The trucks can have different masses M_1 and M_2 . Vehicle 1 hits vehicle 2 at a crash speed V_{2i} .

If drag forces are ignored and only impact forces are included in the analysis then momentum is conserved. Energy is conserved in the elastic collision but not in the plastic collision.

The impact energy is stored in the spring. In the elastic collision the stored impact energy is returned to Vehicle 2. The plastic collision corresponds to the spring energy being captured and used to drive both vehicles forward together. The intermediate behaviour corresponds to vehicle damage (that is, energy absorption) with some recoil. Figure 5 illustrates the simple behaviour.

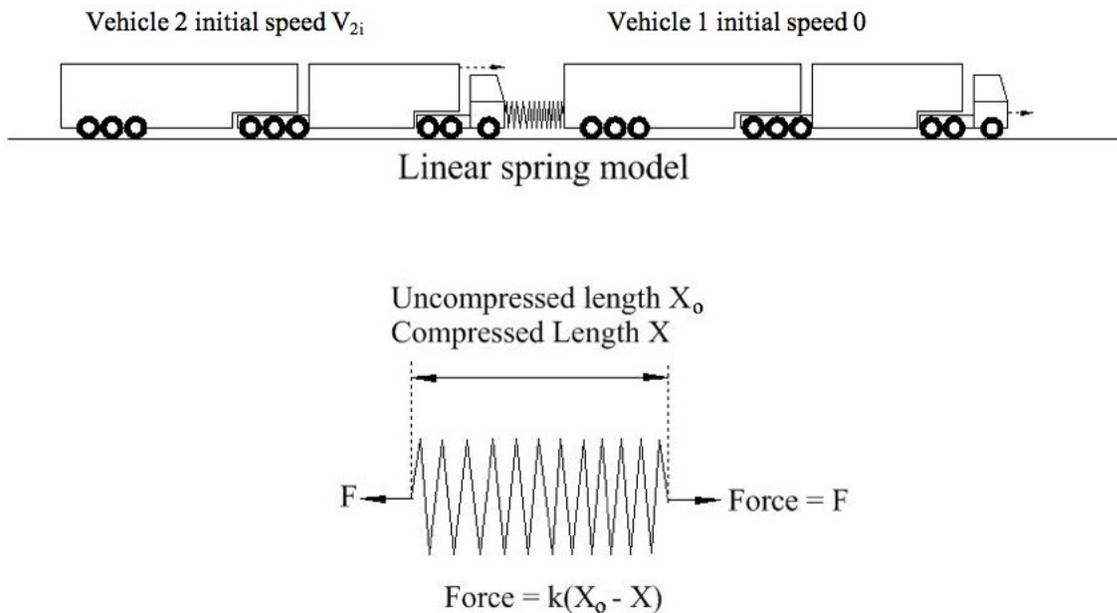


Figure 5 - A simple model of the interaction. The spring has a linear characteristic but can hold (and not return) energy.

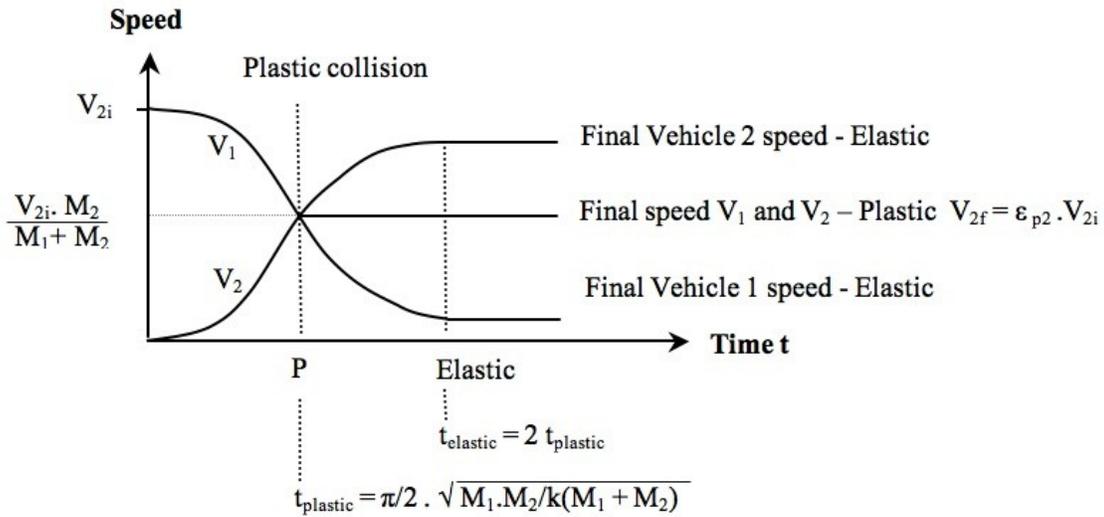


Figure 6 - Speed variation during the collision for a simple spring model where all the impact energy may not be returned.

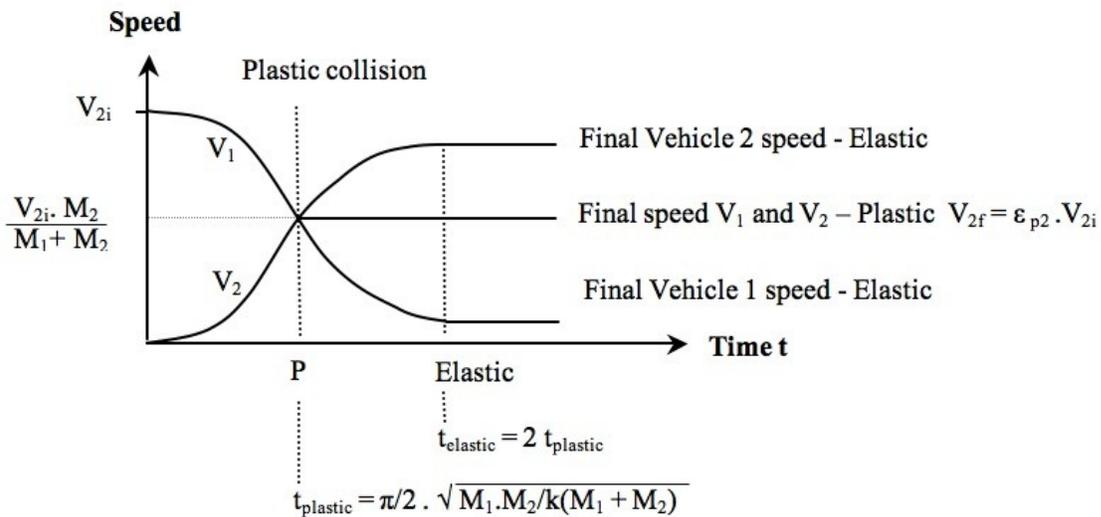


Figure 7 - Acceleration and deceleration variation with time for a linear spring interaction. (P-plastic, E-elastic, I-intermediate, 1-vehicle 1, 2-Vehicle 2)

The deceleration / acceleration solutions for the two vehicles (Figure 7) are:

$$D_1 = - V_{2f} \cdot \sin(\omega t) \cdot \sqrt{k M_2 / M_1(M_1 + M_2)} \quad D_1 \text{ is the deceleration of vehicle 1}$$

$$A_2 = - D_1 \cdot M_1 / M_2 \quad A_2 \text{ is the acceleration of vehicle 2}$$

$$\omega = \sqrt{k (M_1 + M_2) / M_2 \cdot M_1} \quad (7)$$

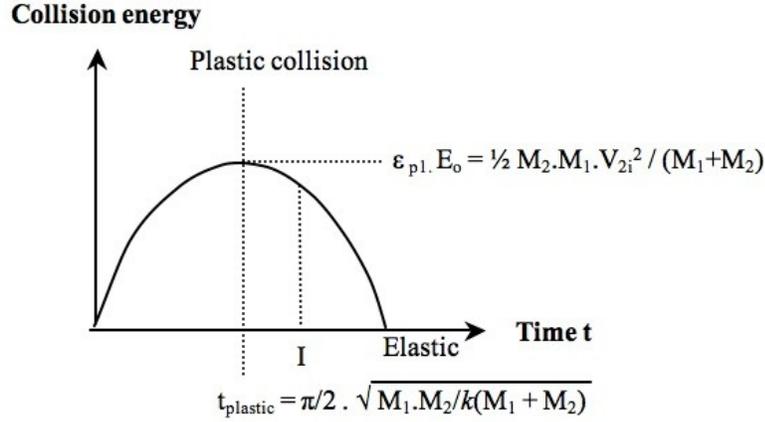


Figure 8 - Variation of the collision energy with time

For a plastic collision:

$$\text{Interaction time is } t_{\text{plastic}} = \pi/2 \sqrt{M_1 \cdot M_2 / k \cdot (M_1 + M_2)} \quad (8)$$

$$V_{2f} = V_{2i} \cdot M_2 / (M_1 + M_2) = \epsilon_{p2} \cdot V_{2i}$$

$$\text{Crush Distance CL} = V_{2i} \cdot \sqrt{M_2 \cdot M_1 / k \cdot (M_1 + M_2)} = 2 \cdot t_{\text{plastic}} \cdot V_{2i} / \pi$$

which corresponds to the maximum compression of the spring (9)

$$\text{Average force} = 1/2 V_{2i} \cdot \sqrt{k \cdot M_2 \cdot M_1 / (M_1 + M_2)} \quad (\text{half the peak value}) \quad (10)$$

$$\text{Energy absorbed} = 1/2 M_1 \cdot V_{2i}^2 \cdot M_2 / (M_1 + M_2) = \epsilon_{p1} \cdot E_o = F_{\text{av}} \cdot \text{CL} \quad (11)$$

In the general case, irrespective of the spring model, the energy-momentum equations show that for vehicle 2:

$$V_2(t) = \epsilon_{p2} V_{2i} \pm V_{2i} \cdot \epsilon_{p1} \sqrt{1 - \epsilon(t) / \epsilon_{p1}} \quad (12)$$

For a plastic collision $\epsilon = \epsilon_{p1}$, Eqn (10) reduces to Eqn (1) and for an elastic collision $\epsilon(\infty) = 0$ and Eqn (10) gives:

$$V_{2f} = \epsilon_{p2} \cdot V_{2i} - V_{2i} \cdot \epsilon_{p1} \quad (13)$$

And forces F_1 & F_2 :

$$F_2(t) = M_2 \, dV_2(t) / dt = - V_{2i} \cdot M_2 \cdot d\epsilon / dt \cdot 1 / \{ 2 \sqrt{1 - \epsilon(t) / \epsilon_{p1}} \} = - F_1(t) \quad (14)$$

In the general case for vehicle 1:

$$V_{1f} = \epsilon_{p2} V_{2i} -/+ V_{2i} \epsilon_{p2} \sqrt{1 - \epsilon(t) / \epsilon_{p1}} \quad (15)$$

The impact interaction time is T

$$T = \text{CL} / \{ V_{2i} \sqrt{1 - \epsilon(\infty) / \epsilon_{p1}} - 1/2 V_{2i} \} \quad (16)$$

Figure 9 illustrates the concepts that arise from the speed and force equations.

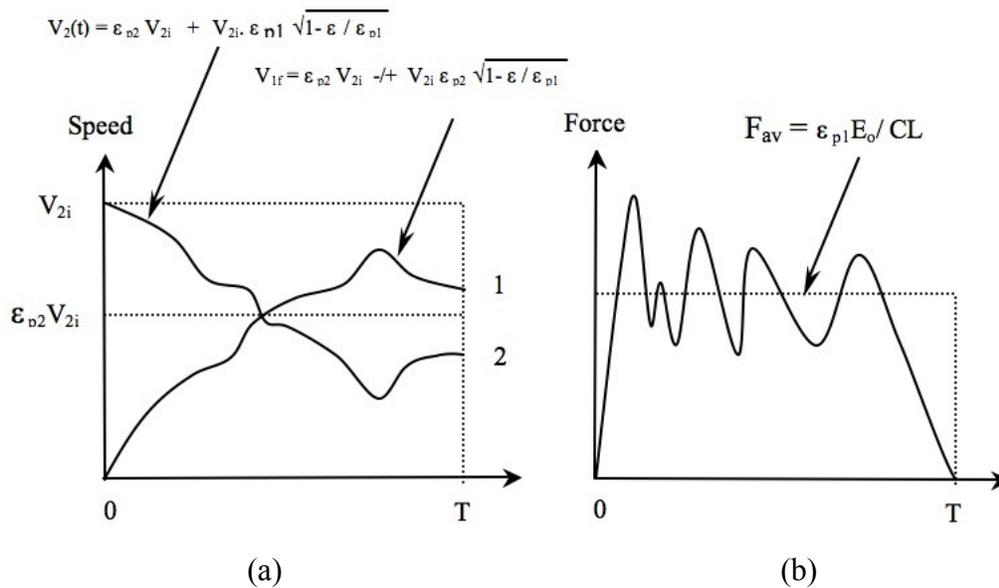


Figure 9 - Speed and force behaviour during a non-idealized crash .
 (a) Speed behaviour (b) Force behaviour

7. Force Dispersion

A heavy truck structure consists of a chassis rail ladder. The longitudinal chassis rails provide the ‘rails’ of the ladder and the cross-members, axles, cabin structure and engine provide the rungs.

A simple model of the structure is shown in Figure 10. The model can be studied as an electrical analog circuit. The transmitted force reduces as a consequence of the both shunt force flow and the dispersive effects of the spring and loss elements. The model is excited with the idealized plastic-collision impulse $F_{av} \times T$.

The model consists of spring elements, dashpots (resistive) losses (dashpots) and masses (capacitance). The masses within each section are lumped into a single mass, which is in the shunt position. The crash forces propagate along the chassis ladder and cause deceleration of each mass element. Each mass element absorbs some force (in proportion to its mass value). The transmitted force in the series elements is transferred to following masses.

It is beyond the scope of this paper to quantify the values of the elements in the analog and therefore the extent by which the transmitted force is dispersed along the model. The point is to highlight that dispersion will reduce the peak force level. The dispersion effect has been ignored in the force calculation (Figure 3) for the truck coupling on vehicle 2.

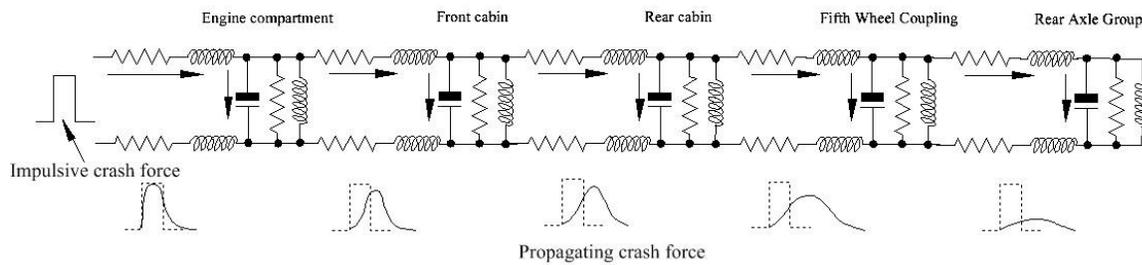


Figure 10 - An electrical analog model for the five-rung chassis ladder model.

8. Real World Collision Considerations

The real-world collision is intermediate between the plastic and elastic collisions. It corresponds to a particular balance of elastic (returned energy) and plastic (absorbed energy) by the spring. The greater is the spring constant k the greater is the deceleration / acceleration and the shorter the interaction time and crush length (CL).

Three relative mass cases can be identified: $M_2 > M_1$, $M_2 = M_1$ and $M_2 < M_1$. Considering an elastic collision, each of the cases must have a time at which $V_1(t) = V_2(t)$. This is the time that the vehicles start to move away from one another during the crash interaction.

The analysis shows that the average crash forces can be estimated if the initial speeds and the final crush length are known. The approach assumes a plastic collision occurs and that the average force is a reasonable approximation to the peak force that occurs.

In reality the interaction is not linear because materials yield or the chassis ladder buckles. Figure 11 illustrates the distinction between the actual behaviour and the linear model approach. The actual situation can be calculated but this requires detailed modeling of the interacting parts of the two trucks.

Dispersion of the transmitted force will reduce the peak impact force when compared to a rigid link. On the other hand, the calculation of an average impact force using the principle of virtual work will underestimate the peak impact force. Therefore this underestimation tends to balance out the dispersion effect.

One consequence of making the front of trucks and the rear of trailers stiffer is that the stiffness k is increased so that the crush length is reduced and the average force is increased. Based on the linear spring model, as trucks become heavier, the forces and

decelerations become greater in proportion to: $\sqrt{k \cdot M_2 \cdot M_1 / (M_1 + M_2)}$

The peak crash forces that occur for a plastic collision are no different to those for the elastic collision. The difference between these collision types is in the length of the interaction and the different final speeds that occur.

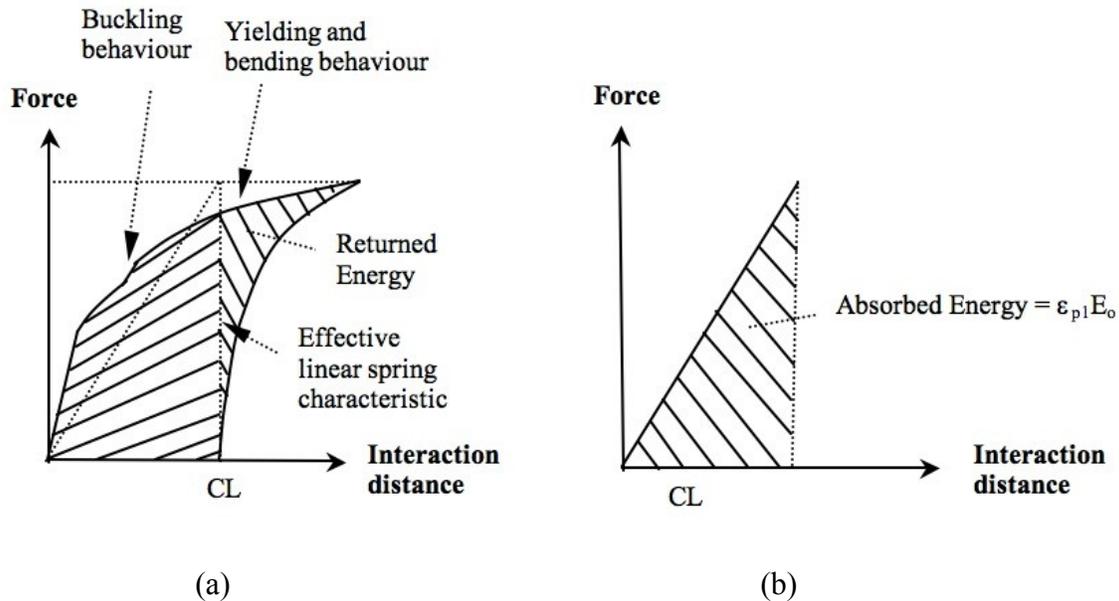


Figure 11 - Illustration of the actual impact situation using a spring-constant approach.
 (a) real situation (b) Linear model without energy return

9. Conclusions

The paper has presented a practical method for estimating the impact forces in the crush zone when trucks collide. It is assumed that the average impact force is the same as that occurring for a ‘plastic’ collision, That is, a collision for which both vehicles move together after the impact.

It has been shown that crash interaction via a linear spring model leads to an elastic collision. If however, the spring energy is dissipated when the two vehicles have the same speed then the model generates the plastic-collision results. This provides the insight that a real-world collision can be viewed as an intermediate case between a plastic and elastic collisions. A real crash exhibits both energy absorption (plastic) and energy storage (elastic) during the interaction.

It is impractical to calculate the crash forces because analysis of the truck structures is very complex and difficult task. In contrast the ‘plastic collision’ method can be applied using spreadsheet-level programming.

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